

Deployment Mechanism for Thermal Pointing System

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Abstract

The Deployment Mechanism for the Total and Spectral Solar Irradiance Sensor (TSIS) is responsible for bringing the Thermal Pointing System (TPS) from its stowed, launch locked position to the on-orbit deployed, operational position. The Deployment Mechanism also provides structural support for the TSIS optical bench and two-axis gimbal. An engineering model of the Deployment Mechanism has been environmentally qualified and life tested. This paper will give an overview of the TSIS mission and then describe the development, design, and testing of the Deployment Mechanism.

Introduction

The goal of the TSIS instrument suite is to accurately measure Total Solar Irradiance (TSI) and Solar Spectral Irradiance (SSI) and make the data available to the research community and public. TSI is defined as the Sun's radiant power per unit area (watts/m^2) incident on a plane surface at the top of the atmosphere that is normal to the direction from the Sun and normalized to 1 astronomical unit. SSI is the power per unit area per unit wavelength interval. These two measurements are important for measuring the magnitude and variability of natural radiative forcing on the Earth's climate system. Radiative forcings are variations in the radiant energy received by the Earth or the energy radiated back to space. Long term measurements of solar and spectral irradiance provided by TSIS are required to continue the Climate Data Record and to fully understand the causes of climate change.

The University of Colorado/Laboratory for Atmospheric and Space Physics (LASP) is responsible for the delivery of the TSIS instrument. The TSIS dual instrument package was originally selected to fly on National Polar-orbiting Operational Environment Satellite System (NPOESS). It was de-manifested in 2006 during the restructuring of NPOESS, and then restored in 2007 following a decision by the NPOESS Executive Committee because of its critical role in determining the natural forcings of the climate system and the high priority given by the Earth Science Decadal Survey. Further restructuring of NPOESS transitioned all climate sensors, including TSIS, to the Joint Polar Satellite System (JPSS). The JPSS is the next generation of low Earth, polar orbiting, environmental satellites that are procured by the National Oceanic and Atmospheric Administration (NOAA) through NASA [1].

TSIS Mission Design

TSIS is comprised of two instruments: the Total Irradiance Monitor (TIM) and the Spectral Irradiance Monitor (SIM). The TIM instrument measures TSI, integrating all wavelengths of solar energy (W/m^2) at the outer boundaries of the atmosphere. The SIM instrument measures SSI from 200 nm to 2400 nm (96% of TSI)[2]. Both instruments along with associated electronics boxes are mounted to an optical bench which is mounted on a two-axis gimbal that provides the required solar pointing accuracy. The two-axis gimbal is mounted inside a mast that can be deployed and stowed (see Figure 1).

The TSIS is slated to launch in 2017 on JPSS Free Flyer 1 (JPSS-FF1), which is a smaller spacecraft relative to NPOESS with solar pointing capability. Since the JPSS-FF1 spacecraft will provide solar pointing, the need for the TPS and Deployment Mechanism is eliminated. Much of the fabrication and assembly of the TPS and Deployment Mechanism was already completed at the time of this decision, so approval was provided to continue the design and testing. This decision was based on the possibility that

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the TPS and Deployment Mechanism may be used on a future mission with similar requirements to the original NPOESS mission. The Deployment Mechanism was successfully built and tested in 2012. Currently an engineering model with increased functionality is being fabricated, and the assembly will be tested in the spring of 2014.

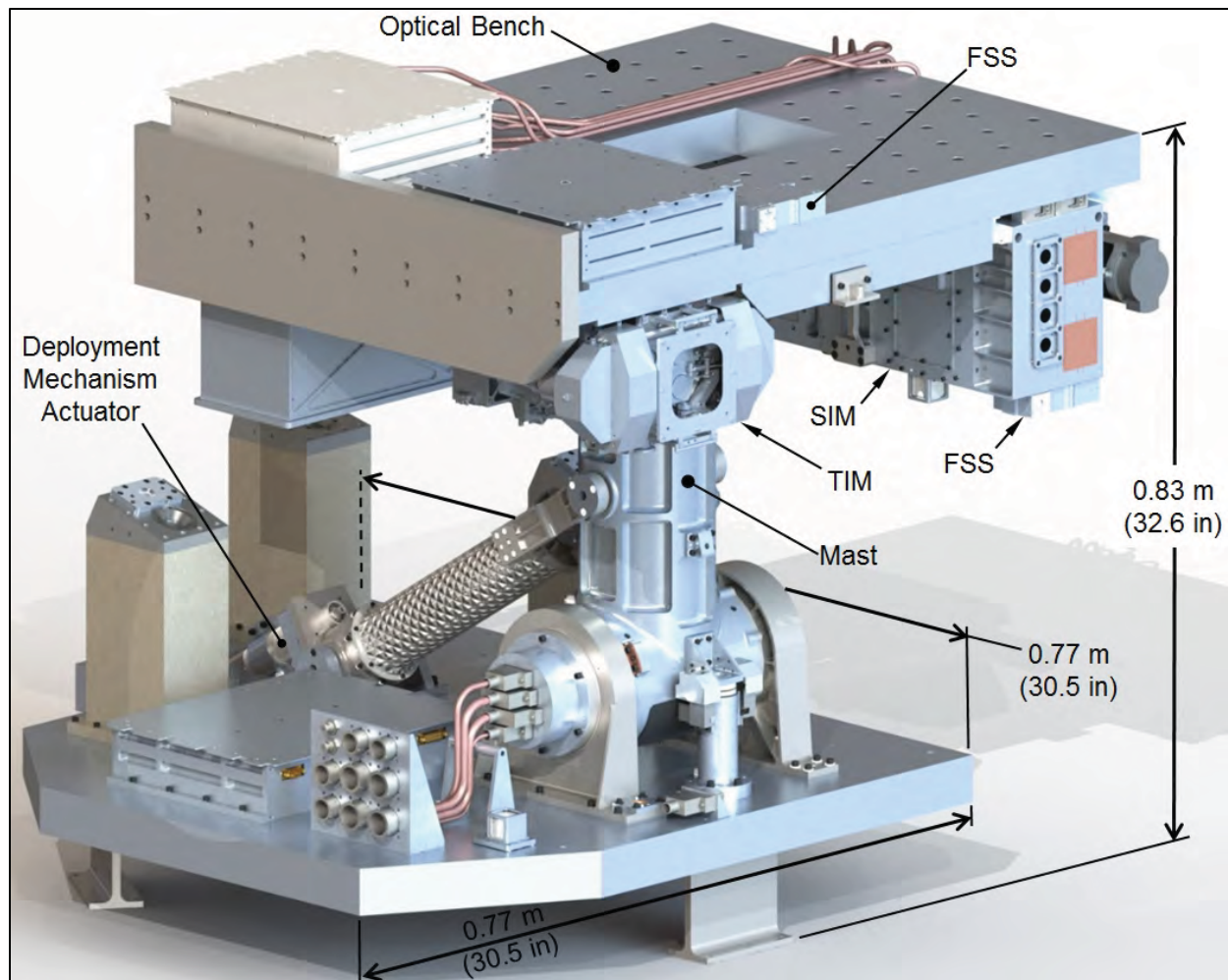


Figure 1. TSIS Rendering

Table 1. TPS Key Driving Requirements

Requirement	Method
Precise tracking of the sun	Gimbale optical bench
Stable thermal environment for optical bench	Material selection, subsystem layout
Steady-state pointing error of TPS with respect to sun center <64 arcsec 1σ	FSS (Fine Sun Sensor), precision actuators, control system
Dynamic pointing error of TPS with respect to sun center <61 arcsec 1σ	Precision actuators, control system
Operational Temperature Range -20°C to $+40^{\circ}\text{C}$	Material selection & platings, design
Survival Temperature Range -30°C to $+50^{\circ}\text{C}$	Material selection & platings, design
Natural frequency of TSIS during launch >50 Hz	Launch latch design, structural design
Natural frequency of TSIS on orbit >6 Hz	Gimbal design, deployment mechanism
TPS shall be operable in any orientation in 1G environment	Gimbal design, deployment mechanism
Uncompensated momentum contribution from TPS $<\pm 0.5$ N-m per axis	Precision actuators, control system, structural design

Thermal Pointing System Design

The TPS was designed to meet the original NPOESS requirements summarized in Table 1. The natural frequency at launch must be >50 Hz and this requires the TPS to have both stowed and deployed configurations. In the stowed configuration, the TPS is locked down to three launch towers with a 2-2-2 kinematic mount. The three attachment points are fastened with ground resettable non-pyro separation nuts. The kinematic restraint uses a vee-spherolinder-cone design (Figure 8) to accommodate launch loads. The TPS is moved from the stowed to the deployed configuration by the Deployment Mechanism (the subject of this paper). Figure 2 illustrates the phases that the TPS goes through after the separation nuts are fired and it is deployed by the Deployment Mechanism.

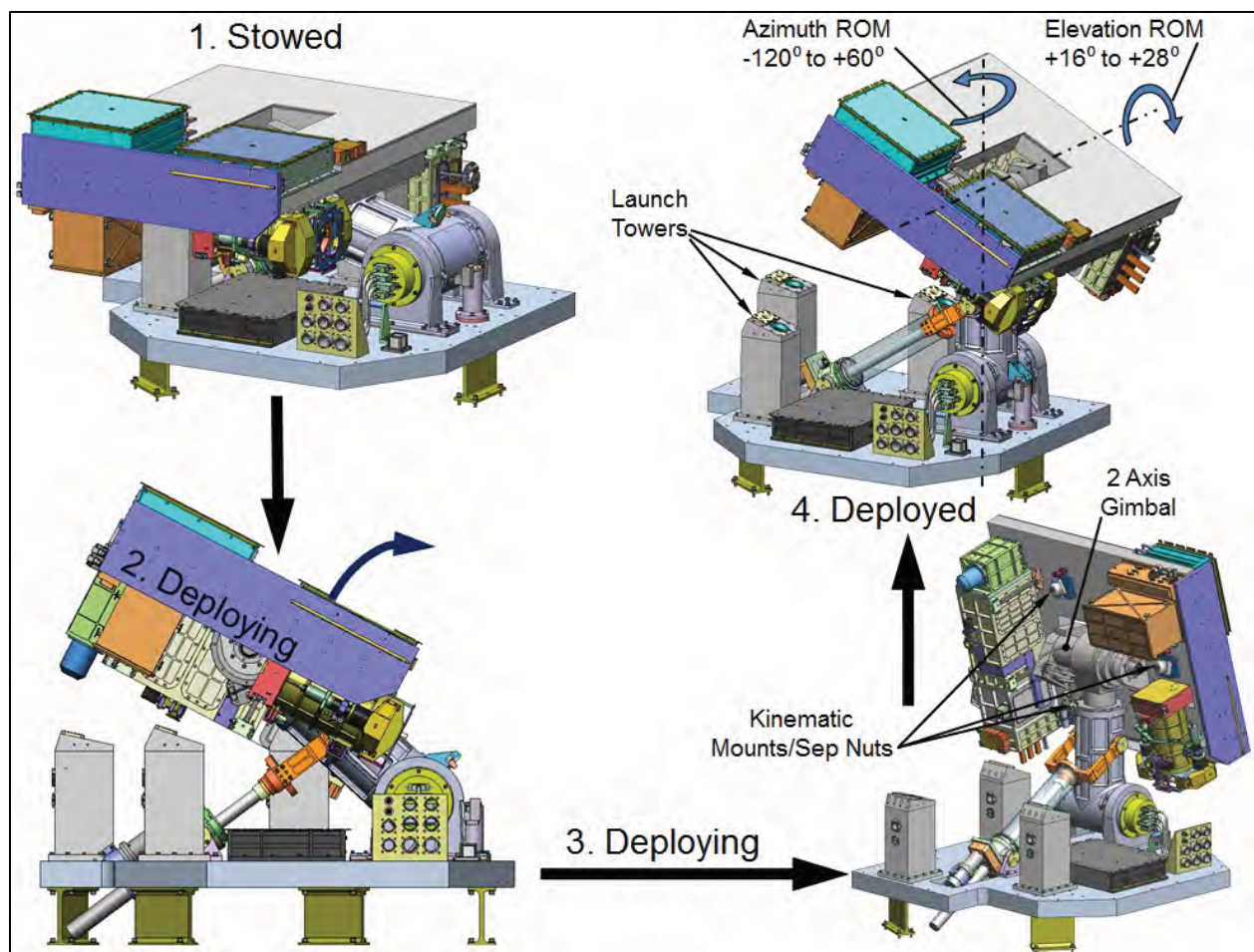


Figure 2. TPS Deployment Phases

Deployment Mechanism Design Overview

The Deployment Mechanism incorporates a unique design that utilizes a variety of moving mechanical components to drive the TPS from stowed to deployed and back again. A stepper gear motor drives the system that includes a ball screw, gears, bearings, flexures, welded bellows and position sensors. The mechanism has a 75° range of motion between the stowed and deployed positions. Redundancy has been included where it is feasible with redundant motor windings and positions sensors. Front and aft views of the Deployment Mechanism engineering model are shown in Figure 3.

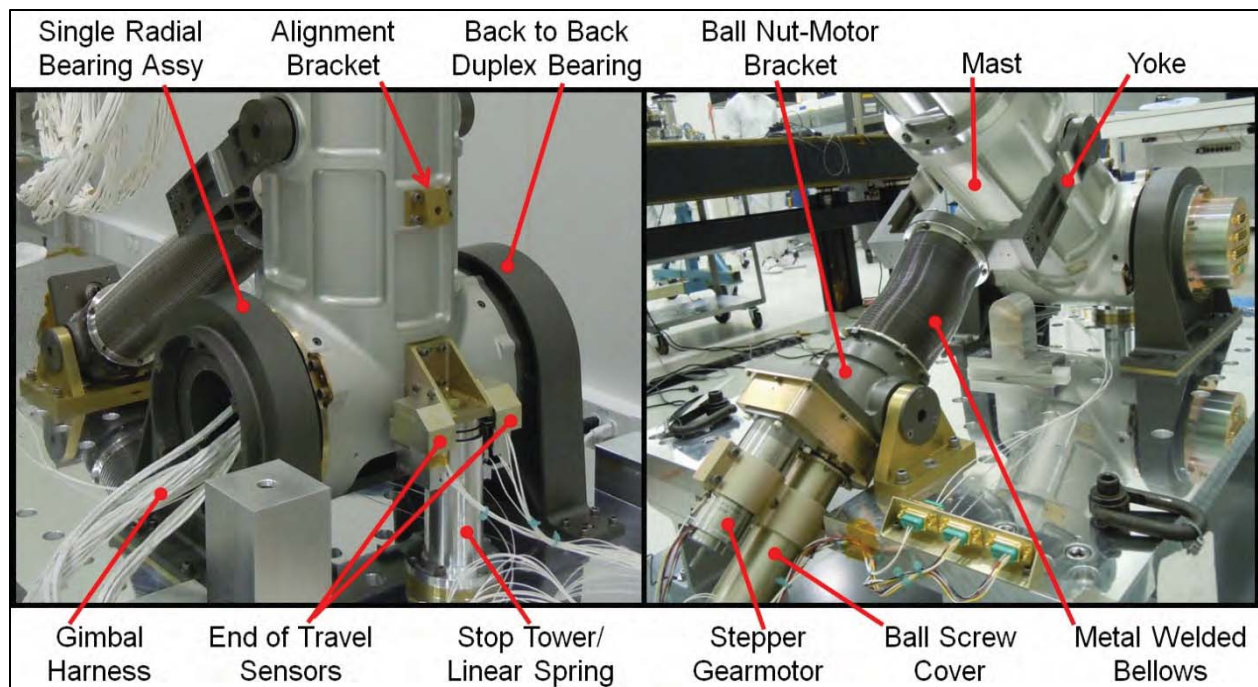


Figure 3. Front and Aft Views of the Deployment Mechanism Engineering Model

Deployment Mechanism Function

The Deployment Mechanism is driven by a 45°, 2-phase stepper motor with an integral 3-stage, 384:1 planetary gear head and a detent brake. The output shaft of the gearmotor attaches to a 2:1 pinion-spur gear reduction as shown in Figures 4 & 7. The spur gear is attached to a ball nut which is housed inside a titanium bracket and is supported by thin section duplex bearings at the top and a radial thin section bearing at the base. The ball nut rotates to advance a 6.35-mm (.25-in) pitch ball screw shaft with 25.4 cm (10 in) of total travel. The top end of the ball screw shaft is attached to a titanium yoke with flexures. A stiff linear spring along the axis of the ball screw shaft at the interface joint with the yoke provides an over travel gage during the stowing operation. The yoke is attached to the TPS mast and pivots on radial bearings at each fork end. The TPS mast is supported by 3rd axis bearings at its base. The 3rd axis bearings include 80-mm (3.15-in) bore duplex back-to-back pair angular contact bearings on one side and an 80-mm bore single radial bearing on the other side. The single radial bearing is preloaded with a beryllium copper flexure.

When the stepper motor drives the ball nut, the mast is driven between the stowed position at 15° and the deployed position at 90°. Optical end of travel sensors indicate when the mechanism is in either the stowed or deployed positions. A stop tower with a stiff linear spring that is compressed when the unit is in the deployed position is used to preload the system. Detent magnets integral to the stepper motor are used to prevent backdriving the system when it is deployed. To prevent the ball screw from over traveling, a non-jamming hard stop engages slightly beyond the nominal deployed position. Figure 4 shows a section view of the Deployment Mechanism CAD model.

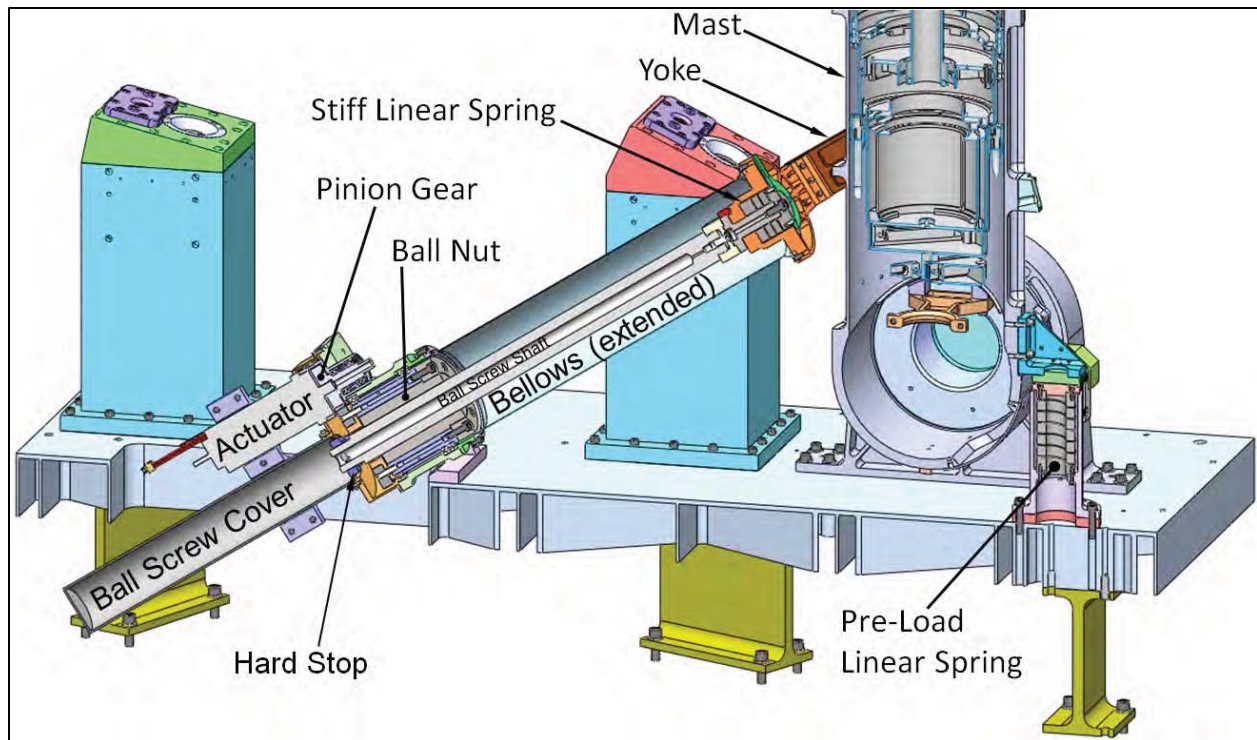


Figure 4. Deployment Mechanism Section View, Deployed Configuration

TPS Deployment Mechanism Design Background

The initial conceptual development for the TPS Deployment Mechanism began in 2007 by LASP engineer Ryan Lewis. The design concept shown in Figure 5 utilized a 7-stage stepper gearmotor to deploy and stow the unit. A latch tower was used to lock the system in the deployed position. Prior to the Critical Design Review (CDR) in 2009, the design evolved and the 7-stage stepper gearmotor was eliminated due to cost, schedule and vendor response concerns. Other design changes that occurred around the CDR time frame are listed below.

1. Ball Screw with yoke attachment to the Mast used to give better mechanical advantage.
2. Smaller and more readily available actuator chosen to drive the ball screw.
3. Latch tower eliminated.
4. Bellows used to contain ball screw shaft lubricant.
5. Stiff compression spring/stop tower compressed in the deployed position to preload the system.
6. Duplex and single radial ball bearings used to support the mast.
7. End of travel sensors and hard stops incorporated into the design.
8. Ability to manually actuate the system when the actuator is removed.
9. Stiff compression spring with over-travel gage added for stowing the system.

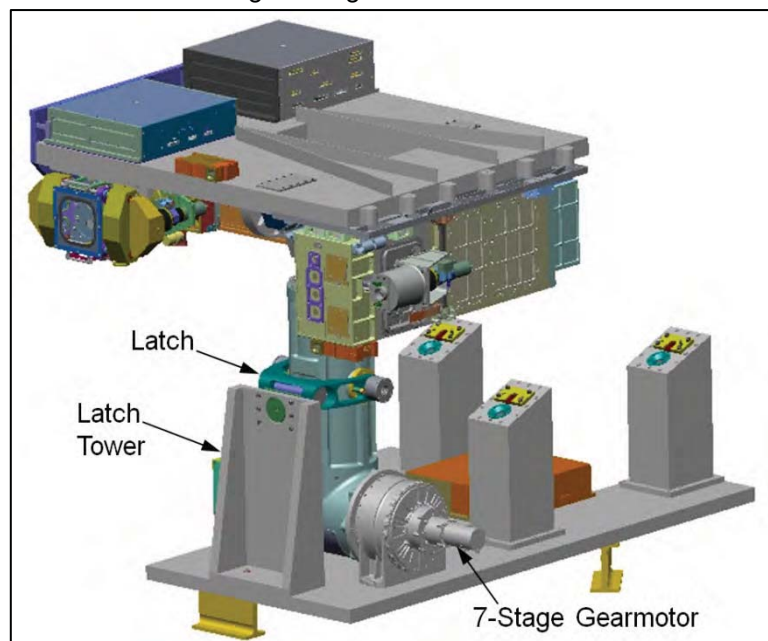


Figure 5. Deployment Mechanism Initial Concept, 2007.

Design Discussion

Motor

The Deployment Mechanism is driven by a 45°, 2-phase and 4-pole stepper motor with redundant windings. This actuator includes a detent brake and a 3-stage, 384:1 planetary gear head integral to the unit. The large gear reduction combined with the 2:1 transfer gears and ball screw results in a large mechanical advantage for the mechanism. The actuator detent torque, 3.4 N·m (30 in·lbf) at the output shaft, combined with the reduced torque reflected back through the ball screw and transfer gears eliminates the need for a latching device in the deployed state.

The 45° step angle equates to 0.12° of rotation at the actuator output shaft per step. The nominal operational rate of the actuator is 128 PPS (Pulses Per Second) which results in an operational speed of 2.5 RPM (Revolutions Per Minute) at the output shaft. At this nominal operational rate, the Deployment Mechanism requires 31.2 minutes to travel from the stowed configuration to the deployed configuration and vice-versa.

The maximum backlash in the gear-motor was measured at <0.0029 radian (<10 arcminutes). This backlash is not an issue for the TPS because when the Deployment Mechanism is in the deployed state, the preload from the stiff compression spring in the stop tower removes all of the play in the system.

The bearings in the motor were lubricated with low vapor pressure Braycote 601EFVB grease and the gears were lubricated with Braycote 601EFVB with 3% MoS₂. All materials in the actuator assembly were chosen to meet the outgassing specification of <1.0% Total Mass Loss (TML) and <0.1% Collected Volatile Condensable Material (CVM).

A thermal clamp bracket is used to conduct heat away from the actuator and into the ball screw cover as shown in Figure 6. Polypad 1000, a 0.23-mm (.009-in) thick thermal conducting material, is compressed between the actuator and clamp to enhance the thermal conductivity between the actuator and ball screw cover.

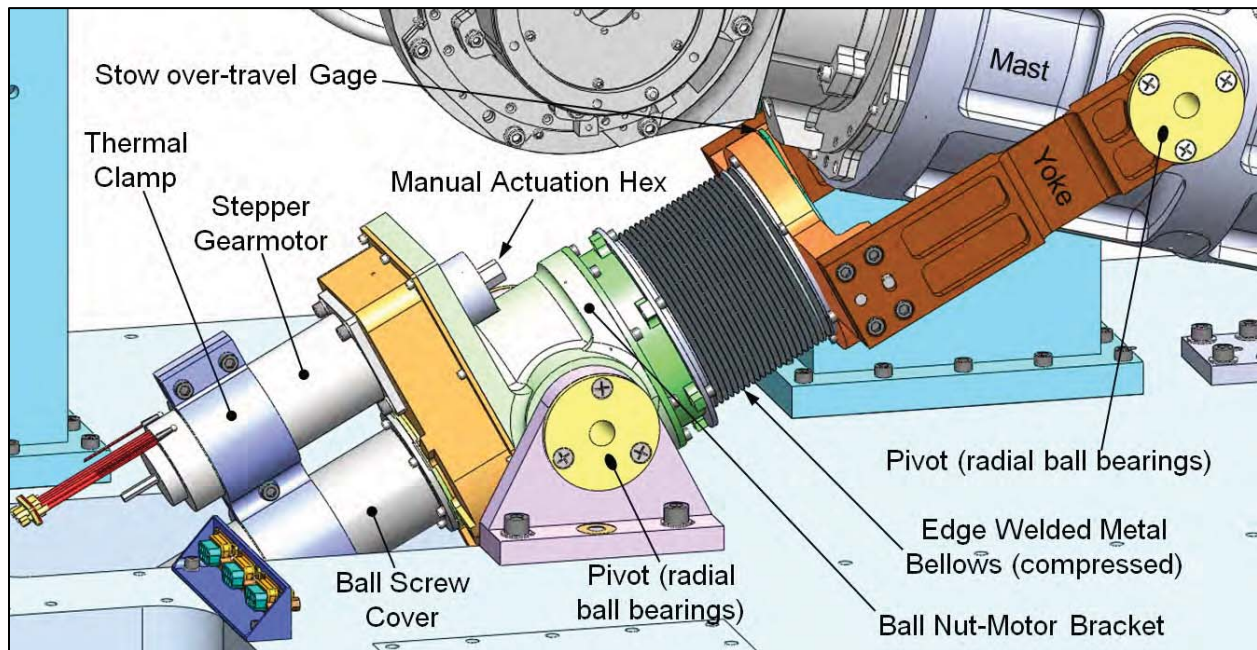


Figure 6. Deployment Mechanism Drive Components

On orbit, the Deployment Mechanism will only be operated one time, moving the TPS from the stowed to deployed configurations. During this one-time deploy sequence the Deployment Mechanism is open-loop controlled. The actuator is commanded to move in the deploy direction and then it will automatically be commanded to stop when either one of the redundant deploy position sensors is triggered.

On the ground during Integration and Testing (I&T), the deploy command sequence is closed-loop controlled, identical to what it is in space. Stowing on the ground is open-loop. The actuator is commanded to move in the stow direction until it is manually commanded to stop slightly prior to reaching the stow position. The actuator is then incrementally commanded to approach the stowed position so that the kinematic interfaces of the 3X vee-spherolinder-cones can be closely monitored and dialed-in for optimal positioning and a flush, slightly preloaded interface.

Transfer Gears

Both the pinion and spur gear have a 20 diametral pitch and 20° pressure angle. The 28-tooth pinion gear is a custom part fabricated from 15-5 PH stainless steel and heat treated to condition H1000. This pinion gear part includes an integral precision shaft for two radial bearings to slip fit over. There is also a 8-mm (5/16-in) hex head at the end of the integral shaft that allows for the Deployment Mechanism to be manually stowed or deployed without an actuator. The 56-tooth spur gear is a standard off-the-shelf part made from 303 stainless steel. The spur gear is modified to add mounting holes, a precision central through hole and a dowel pin slot. Both gears were lubricated with Braycote 601EFVB. There is a small amount of backlash [<0.0026 radian (<9 arcminutes)] in the transfer gears, but as was stated in the motor section above, this is not a concern due to the preload in the deployed state that removes all play in the system.

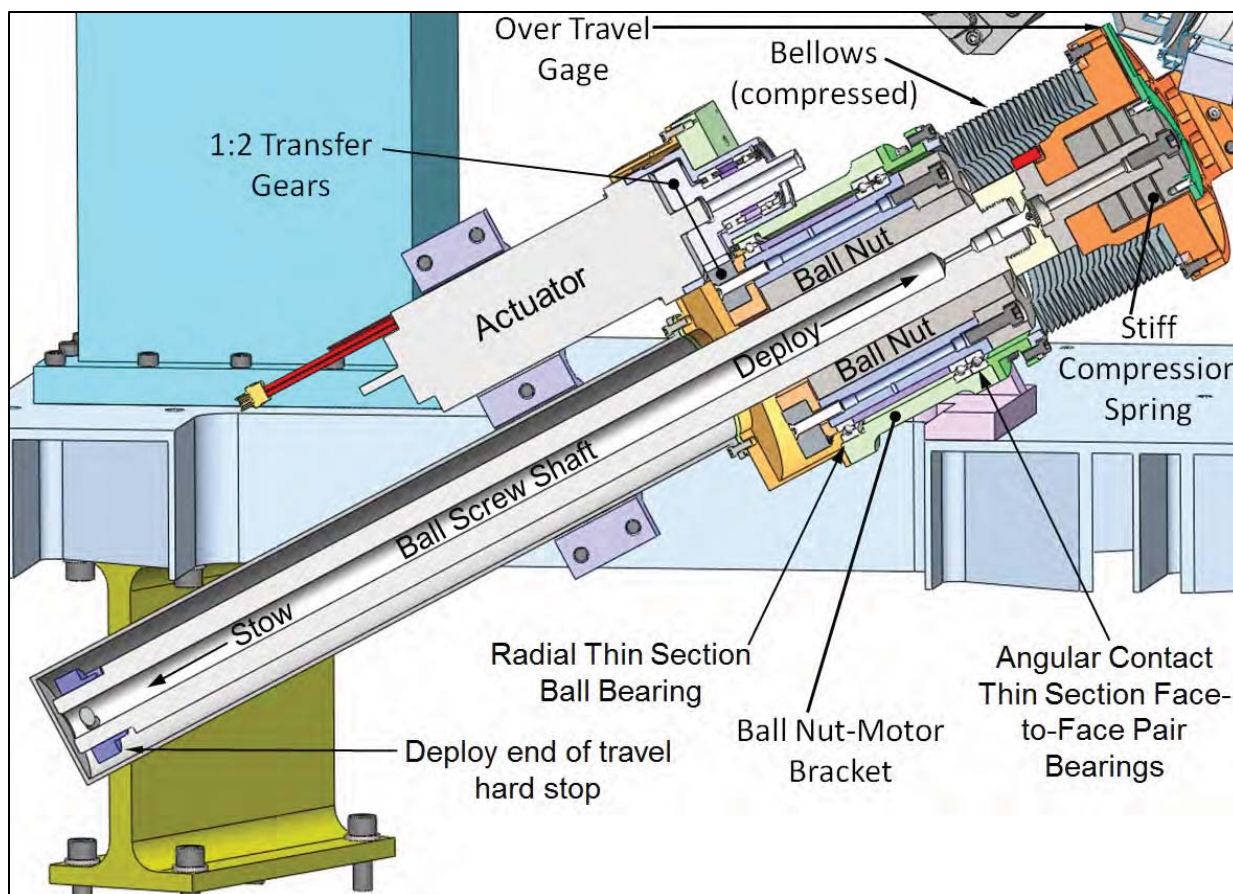


Figure 7. Focus Mechanism Drive Components, Section View

Ball Screw/Nut

The right handed ball screw has a 25.4-mm (1-in) diameter shaft and 6.35-mm (0.25-in) pitch. The ball nut has two tracks/return tubes with 2.5 circuits per track. The ball screw was sized to handle worst case mechanism loads with positive margin. The ball nut is not preloaded and the nut-to-screw backlash is .13-.38 mm (.005-.015 in). As stated earlier, the backlash is removed from the system when the mechanism is in the deployed state due to the preload imparted by the stiff compression spring in the stop tower. The ball screw shaft is fabricated from 4150 alloy steel Rc 56-60 and is chrome plated to .005-.010 mm (.0002-.0004 in) thick. The ball nut is made from AISI 1117, nitrided to Rc 56-60 and electroless nickel coated to .005-.010 mm (.0002-.0004 in) thick. The balls have a nominal diameter of 3.96 mm (.156 in) and are made from AISI 52100 steel. The ball screw shaft grooves, nut and balls are lubricated with Braycote 601EFVB. Barrier film is applied to each end of the ball screw shaft to prevent lubricant migration.

All of the critical drive components of the Deployment Mechanism are brought together by the ball nut-motor bracket. The section view in Figure 7 illustrates how the actuator, transfer gear and ball nut interface with the ball nut-motor bracket. This bracket is fabricated from Titanium 6AL-4V to reduce the CTE mismatch between the ball nut, thin section bearings and bracket. The ball nut rotates on thin section bearings within the bracket when it is driven by the actuator through the transfer gears. The ball screw shaft does not rotate as it advances either in the deployed or stow direction. Throughout the deployment the ball nut-motor bracket pivots on two radial ball bearings that are located in pockets machined into each side of the bracket. The ball nut-motor bracket is attached to a base bracket that is mounted to the baseplate as shown in Figure 6.

Thin Section Bearings

The thin section bearings that support the ball nut are an angular contact face-to-face pair and a single floating radial bearing. The face-to-face style angular contact bearing pair was chosen for their ability to withstand high axial loads. The main load path of the mechanism travels along the axis of the ball screw and through this face-to-face bearing pair. The bearing pair was sized to handle the maximum thrust load with positive margin. The single radial bearing is floating axially and only provides radial support to the ball nut. The load and lifetime cycle requirements make preloading this bearing unnecessary. Both the angular contact bearing pair and single radial bearing are lubricated with Braycote 601EF. The material of the thin section bearing races and balls is 440C stainless steel.

Bellows

The expandable and collapsible bellows is used to contain the lubricant on the ball screw shaft. The completed assembly has a compressed length of 4.8 cm (1.9 in) and it can be expanded to 31.5 cm (12.4 in) resulting in a 26.7-cm (10.5-in) stroke capacity which meets the requirements of the Deployment Mechanism. The bellows is fabricated using pieces of .10-mm (.004-in) thick AM350 stainless steel that are hydraulically stamped into diaphragms. Once stamped, several of the diaphragms

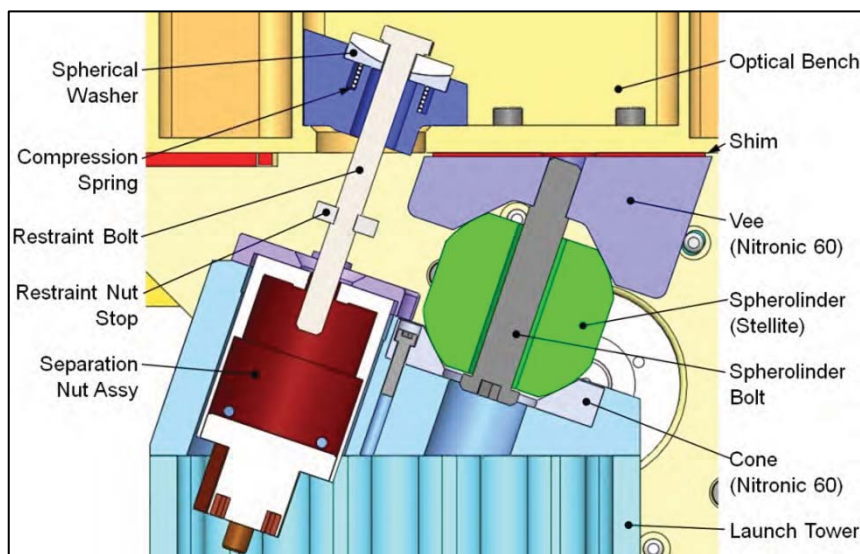


Figure 8. Kinematic Mounts/Restraint Bolt Section View in the Launch Configuration

are placed back-to-back and welded together at both the inside and outside edges. Custom 316L stainless steel mounting flanges are then welded to each end of the bellows. The spring rate of the completed bellows is 670 N/m (3.8 lbf/in) and it has a rated cycle life of 100,000 cycles. Initially, there was a concern that the bellows might sag, buckle or squirm during the deploy/stow cycles and thereby interfere with the ball screw. The buckling and squirming issues were eliminated by venting the bellows (the bellows does not need to hold a vacuum or pressure). The bellows has only minimal sag in the deployed state such that the inside of the bellows does not touch the shaft of the ball screw at any time during the deployment.

Stow Over-Travel Gage

During the stow operation, the actuator must drive the mast down so that the 3X kinematic vee-spherolinder-cones are in flush contact with each other (see Figure 8). Only at that point can the restraint bolts be torqued for launch configuration. An over-travel gage has been designed into the system to avoid over driving the ball nut once nominal kinematic engagement has been reached. This gage includes a stiff linear spring [2,600 N/mm (15,000 lbf/in)] attached to the end of the ball screw shaft along with a simple aluminum gage mounted to the top of the spring as shown in Figure 7. The nominal gap between the aluminum gage and the yoke base is 1 mm (.04 in). If the ball nut is driven past nominal engagement, this gap will decrease. Monitoring this gap during the stowing operation is essential to avoid over-driving the ball nut.

Hard Stop

The deployed hard stop is used to avoid damaging the ball nut/screw in the unlikely event that both end-of-travel position sensors fail during deployment. The hard stop uses a radial face-to-face design that is non-jamming and minimizes stress on the ball screw. During the initial build of the Deployment Mechanism, the hard stop was positioned onto the ball screw such that it engaged with a stop tab on the ball nut at a position slightly beyond the nominal deployed position. In this nominal position the stop tower compression spring is displaced by 2.54 mm (.100 in). The hard stop engages when the stop tower compression spring is displaced by 3.18 mm (.125 in). During normal operation, the end-of-travel will be reached before the hard stop will engage. Figure 9 shows the hard stop when it is engaged. After the hard stop was correctly positioned, the ball screw was removed from the assembly and the hard stop was match drilled to it. During this operation the threads of the ball screw were completely covered to prevent contamination. A screw and a dowel pin were used to fasten the two pieces together. The ball screw and hard stop were then reassembled into the mechanism after the match drilling operation.

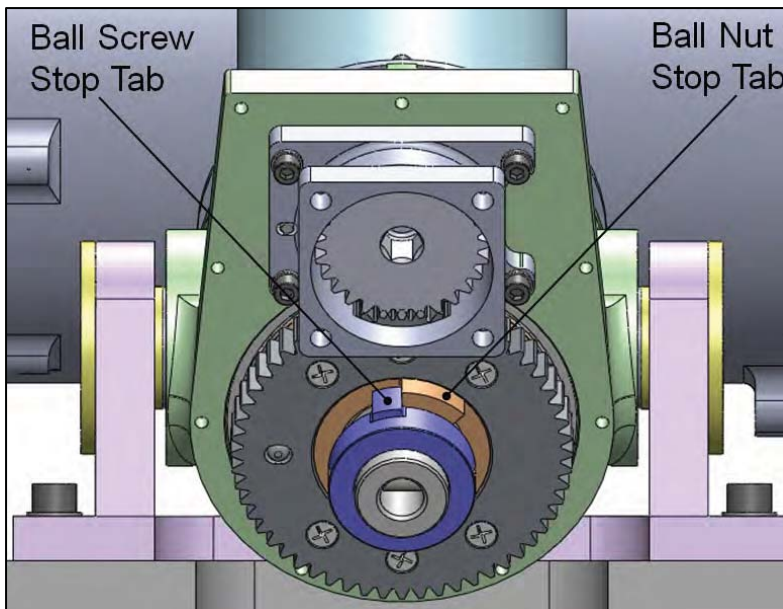


Figure 9. Deployed Hard Stop

Yoke to Mast Interface

The titanium yoke is attached to the aluminum mast via two radial ball bearings that fit into pockets machined into the side of the mast as shown in Figure 11. A titanium ring is placed around each 440C ball bearing to prevent damaging the ball bearings during cold survival due to the CTE mismatch between the aluminum mast and 440C bearing rings. Each yoke arm has a single blade flexure machined into it to allow for some compliance in the system.

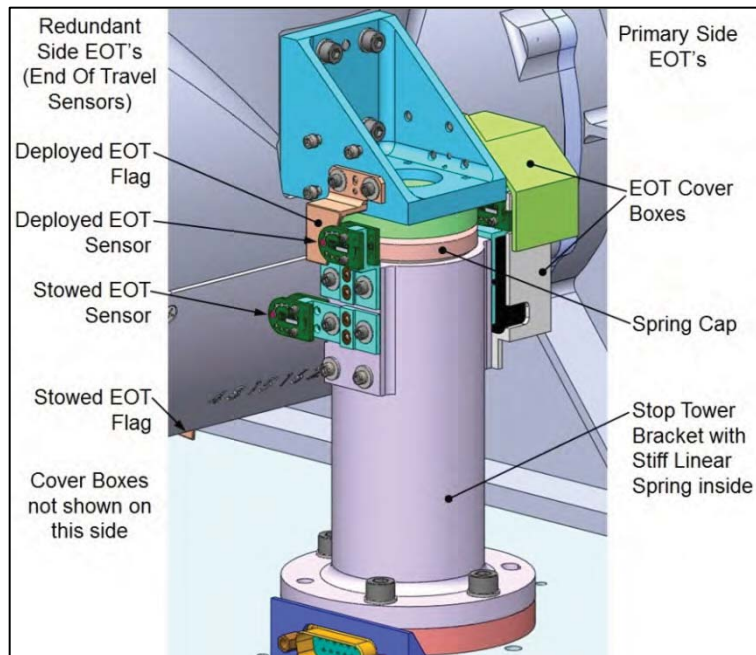


Figure 10. Stop Tower and End of Travel Sensors

Stop Tower and End of Travel Sensors

The stop tower assembly is used to preload the system in the deployed state. This is accomplished with a stiff compression spring that is mounted inside the stop tower bracket. As was mentioned earlier, the spring is compressed by 2.54 mm (.100 in) in the nominal deployed position, which results in a force of 1,628 N (366 lbf). When this force is opposed by the ball screw, all backlash in the system is removed. This results in a stiff design that meets the solar tracking requirements of >6 Hz in the deployed configuration. Redundant end-of-travel sensors are also mounted to the stop tower bracket. These sensors are phototransistor/infrared emitting diode pairs and their positions are dialed into the ideal spot during the initial assembly of the Deployment Mechanism. Interrupt flags are mounted

to the mast as shown in Figure 10. Cover boxes are mounted around the sensors to prevent external light from potentially causing sensor reading errors.

3rd Axis Bearings

The 3rd axis bearings that support the mast assembly (see Figures 3 and 11) are comprised of an angular contact duplex pair on one side and a single radial ball bearing on the opposite side. The duplex back-to-back pair was chosen for excellent radial, axial and moment stability. The bearing pair size is: 80-mm bore x 125-mm outside diameter x 22-mm wide and there is a 311-N (70-lbf) preload. The single radial bearing size is: 80-mm bore x 140-mm outside diameter x 26-mm wide. This bearing has an axial preload of 44.5 N (10 lbf) that is applied with

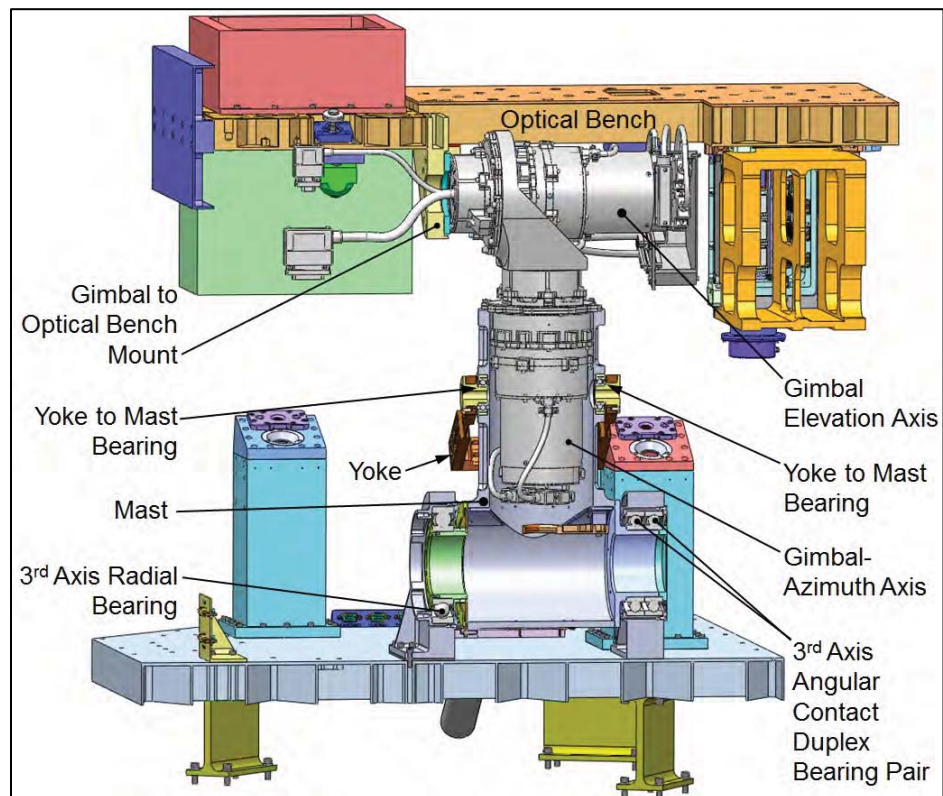


Figure 11. Deployment Mechanism Vertical Section View

a beryllium copper diaphragm flexure to maintain ball-race contact and prevent skidding. The 3rd axis bearings were sized to handle worst case loads with positive margins. The support brackets were fabricated from Titanium 6AL-4V for excellent strength and stiffness characteristics plus low CTE to prevent damaging the bearings during cold survival temperatures. The bearings were lubricated with Braycote 601EFVB. Labyrinth seals and barrier film are used to prevent lubrication migration.

Gimbal

The 2-axis gimbal is mounted to the top of the mast with the azimuth axis actuator located mostly inside the vertical shaft of the mast as shown in Figure 11. The optical bench is mounted to the elevation axis actuator via a gimbal to bench mounting bracket. The gimbal harness is routed through the bottom of the mast and out of the 3rd axis single radial bearing bore.

Analysis

The major components of the TPS were designed and analyzed to meet the requirements of the NPOESS mission. ANSYS Workbench was used to predict structural and thermal stresses along with modes during the launch and deployed configurations. The predicted 1st mode is 67 Hz in the stowed configuration and 8.1 Hz in the deployed configuration. Individual piece parts and small sub-assemblies that are in the load path were analyzed in ANSYS and the geometry was modified as required to result in positive margins of safety for worst case load configurations. All moving parts including the actuator, bearings, gears and ball screw were analyzed and sized to withstand lifetime cycles. The mean Hertz contact stress was calculated for all bearings in the system to verify that they met the requirement of <400 ksi (<2760 MPa). All had healthy margins of safety.

Testing

Life Test

In early 2012, the TPS successfully completed a life test of 100 stow-deploy cycles. A mass model that matched the mass and Center of Gravity (CG) of the populated optical bench was attached to the top of the mast for this test. Fifteen of the cycles occurred in the thermal vacuum chamber with 5 cycles at ambient temperature, 5 cycles at -20C, and 5 cycles at +40C. The remaining 85 cycles took place at ambient temperature and pressure in a cleanroom. Each cycle required just over 60 minutes to complete. Figure 12 shows the TPS about to enter the thermal vacuum chamber. Figure 13 shows life test data for cycles 34-40.

During the 100 cycle life test, the actuator operated nearly continuously for a total of 100 hours. A thermocouple was bonded to the body of the motor and the temperature was continuously monitored throughout the entire test. An automatic shut off set at +150C was built into the control software to avoid damaging the actuator. The actuator temperature limits specified by the manufacturer are -80C to +225C. The specified temperature limits for Braycote 601EFVB are -80C to +204C. The temperature gradient between the external surface of the actuator and the thermocouple along with a healthy safety factor were the rationale behind the +150C shut off. The automatic shut off was never triggered during the life test. The maximum temperature that the thermocouple measured was +134C during the hot (+40C) test in the thermal vacuum chamber.

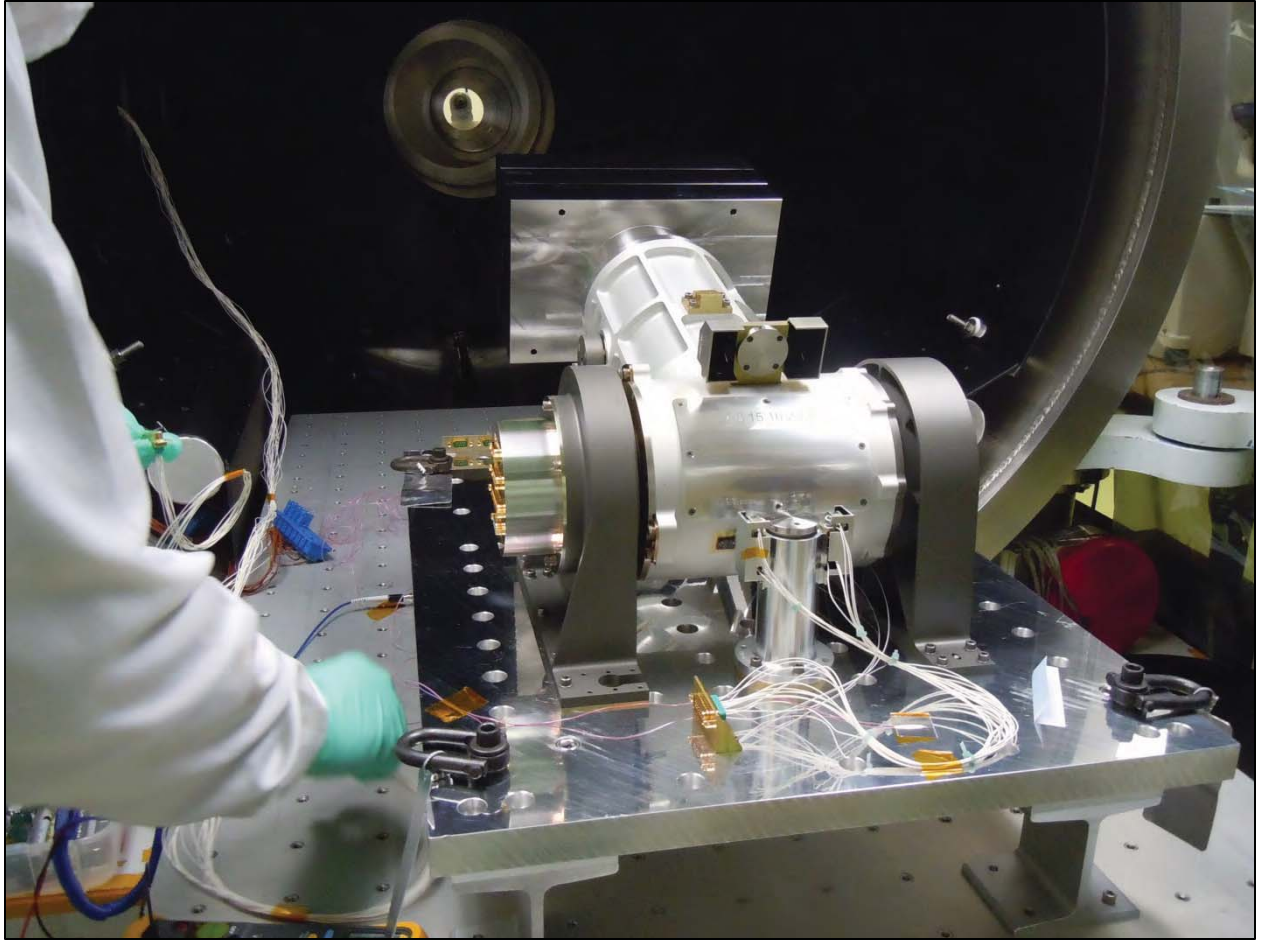


Figure 12. Deployment Mechanism entering the Thermal Vacuum Chamber during the Life Test

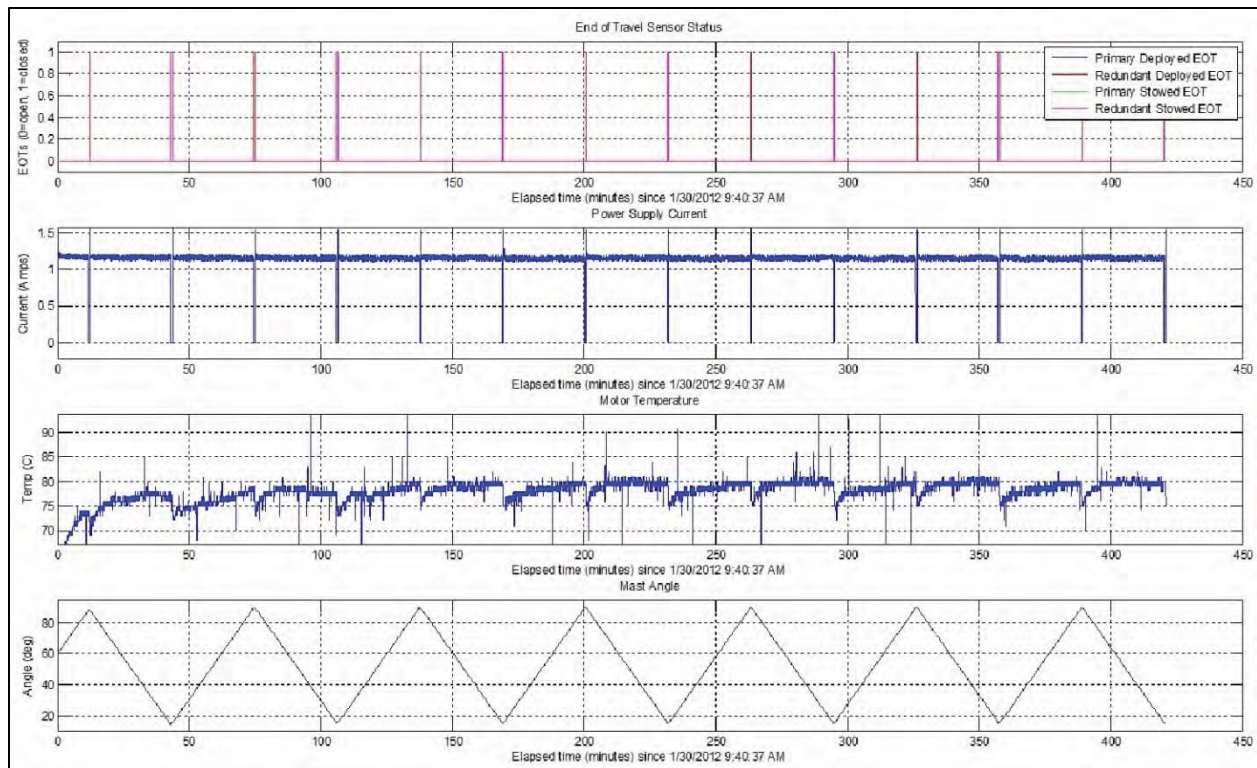


Figure 13. Deployment Mechanism Life Test Data for Cycles 34-40

The maximum current observed during the life test was 1.2 amps at the nominal operation voltage of 28 volts. This correlates to 0.89 N·m (7.9 in·lbf) of torque required by the actuator to stow or deploy the mechanism. This number was calculated using the actuator's torque constant of 26.8 mN·m (3.8 in·oz/watt). During qualification testing at the vendor prior to shipment,

the actuator pull-in torque at

the nominal rate of 128 PPS

was measured to be 14.1 N·m (125 in·lbf), see Figure 14. From these measurements, and using a safety factor of 1.5, the actuator torque margin was calculated to be +9.4.

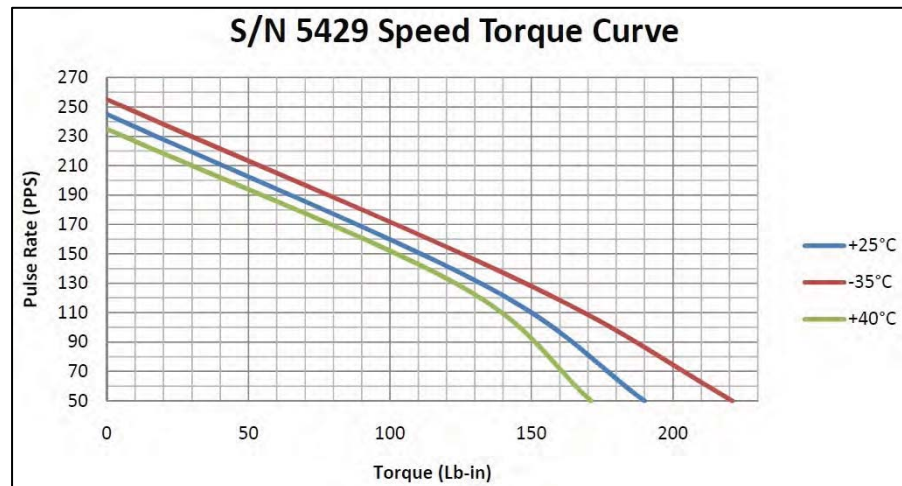


Figure 14. Actuator Speed-Torque Curve at Various Temperatures

Backdriving Test

In early 2012, shortly after the life test, the Deployment Mechanism successfully passed the backdriving test. This test verified that the Deployment Mechanism is able to deploy and stow in any orientation in a 1G environment. A Flotron was used to perform this test as shown in Figure 15. The Deployment Mechanism had healthy torque margin while deploying and stowing in all orientations.

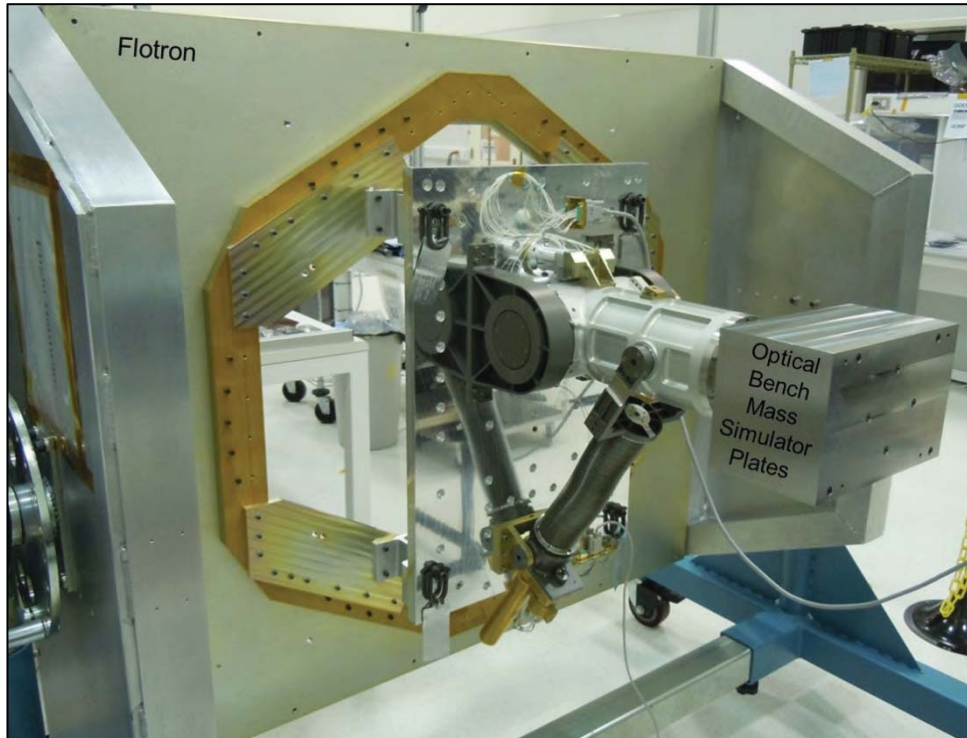


Figure 15. Deployment Mechanism Back Driving Test on the Flotron

The backdriving test also verified that the TPS remains in a stable position in the worst case vertical orientation due to gravity. This was done by measuring the gap between the spring cap and the stop tower bracket (see Figure 10) while the TPS was in the deployed configuration. The gap remained unchanged after 24 hours, thus verifying that the system did not back drive and that the detent torque in the motor was appropriately sized.

The torsional spring rate of the Deployment Mechanism system in the stowed configuration was calculated during this test. During the test, gap measurements between the mast and the baseplate were taken at various times and it was noticed that the gap would change slightly after the Flotron was rotated to a new position. This gap change was the result of the gravitational force acting on the optical bench mass simulator mounted to the top of the mast as shown in Figure 15. Certain orientations resulted in more torque imparted to the mast rotation axis compared to other orientations, thus resulting in a smaller or larger gap. The angle between the mast vertical axis and the baseplate is easily calculated from this gap measurement.

The torsional spring rate in the stowed configuration was calculated with the Flotron in the horizontal position. A gap measurement verified that the mast angle was at the nominal 15° . At this point the optical bench mass simulator plates were removed and the gap was measured again. This time the gap increased by 1.14 mm (.045 in) which correlates to the mast angle increasing to 15.386° . Using this change in angle measurement along the moment torque caused by the mass simulator plates, the torsional spring constant of the Deployment Mechanism system in the stowed configuration was calculated to be 674 N·m (5,967 in·lbf/degree).

Future Testing

A gimbal characterization test will be performed early in 2014 with an optical bench mounted to the gimbal and representative mass models mounted to the optical bench. This test will characterize the gimbal pointing performance when it is mounted to the TPS. Following that test, the engineering model assembly will be expanded to include the launch towers and resettable separation nuts. This unit will then undergo tests to verify deployment and stow/alignment capability. A modal test is also planned in both stow and deployed configurations and the results will be correlated to the structural model.

Conclusions

The Deployment Mechanism design has been proven to be a viable method for deploying and stowing the TPS. The mechanism successfully passed the lifetime and back driving tests. Future testing will undoubtedly reveal more nuances about the design and its features.

Lessons Learned

During the life test, the actuator thermocouple produced transient spikes that exceeded the +150C temperature limit, setting off alarms in the software that resulted in the test automatically stopping and causing delays. This was fixed by increasing the thermocouple sample rate to 200 msec and averaging 5 samples.

The peer review process is extremely helpful in producing a successful design. Prior to CDR, two peer reviews were held at LASP with many LASP mechanical, electrical and systems engineers in the audience. The insightful questions from the audience and the action items that came out of the reviews resulted in improvements to the design. Because of these design improvements the fabrication, assembly and testing of the assembly went smoothly without any major problems.

Acknowledgements

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